

Effect of finite heat input on the power performance of micro heat engines

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ABSTRACT

Micro heat engines have attracted considerable interest in recent years for their potential exploitation as micro power sources in microsystems and portable devices. Thermodynamic modeling can predict the theoretical performance that can be potentially achieved by micro heat engine designs. An appropriate model can not only provide key information at the design stage but also indicate the potential room for improvement in existing micro heat engines. However, there are few models reported to date which are suitable for evaluating the power performance of micro heat engines. This paper presents a new thermodynamic model for determining the theoretical limit of power performance of micro heat engines with consideration to finite heat input and heat leakage. By matching the model components to those of a representative heat engine layout, the theoretical power, power density, and thermal efficiency achievable for a micro heat engine can be obtained for a given set of design parameters. The effects of key design parameters such as length and thermal conductivity of the engine material on these theoretical outputs are also investigated. Possible trade-offs among these performance objectives are discussed. Performance results derived from the developed model are compared with those of a working micro heat engine (P3) as an example.

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1. Introduction

There is an ever increasing demand for power sources on the micron scale which are essential for powering a wide range of microsensors, microactuators, and other applications [1]. In contrast to macro scale power sources, micro power supplies are compact, lightweight, and promise to deliver high power density [2]. Thus, in recent years, there has been significant research worldwide in the development of a variety of micro power generation systems [3,4]. Among these systems, micro heat engines attract major interest as they can be fabricated using mainstream MEMS processes and thus potentially integrated with a wide variety of microsystems [4]. In addition, the combustion of fuel in micro heat engines results in high energy density power sources even when heat engine conversion losses are taken into account [5]. Typical micro heat engine development reported to date includes the MIT micro gas turbine [6], the P3 micro engine [7], micro Wankel engine [8], micro steam engine [9], and micro reciprocating engines [10].

Similar to that of their macro scale counterparts, the overall performance of micro heat engines is determined by tradeoffs of a number of parameters which need to be considered at the design stage. Thermodynamic modeling has been used to predict theoretical limits, e.g. Carnot limit of maximum thermal efficiency in general heat engines. While it is unlikely that real micro heat engines will operate at the theoretical limits predicted by thermodynamics, evaluation of the thermodynamic limit is important as a measure of the potential room for improvement in real-life micro heat engines. Furthermore, theoretical models can also be used to evaluate potential performance with different design parameters. In particular, the down-scaling effect for regenerative heat engines in terms of thermal efficiency has been reported [11]. However, to the best of our knowledge, there are no systematic modeling studies on the theoretical limit of power and power density of micro heat engines as functions of length scale and material selection. Moreover, as thermal isolation of the hot and cold sections of an engine becomes increasingly challenging at the micron scale [12], heat leakage is thus one of the key considerations in micro engine designs [13]. Previous studies in miniaturized heat engines have been based on a thermodynamic heat engine model with leakage [14], as shown in Fig. 1(a), where the relevant parameters are listed in Table 1. Although the thermal efficiency vs. leakage characteristic of a micro heat engine was derived from this

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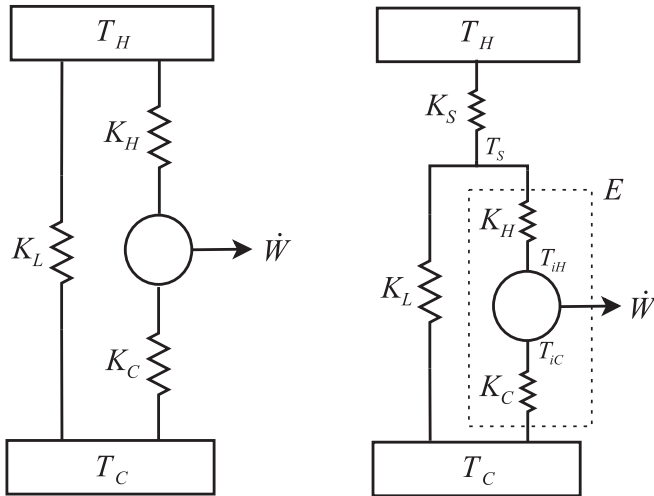


Fig. 1. (a) Thermodynamic model of heat engine with leakage (b) New finite heat model of micro heat engine.

model, the effect of leakage on power generation was not predicted [11]. This is because an infinite amount of input heat Q_H was assumed to be drawn from the source T_H , leading to generated power being independent from leakage [15]. However, in real heat engines, there is a limit in the heat flux that can be drawn from a source [16], which is particularly important for micro heat engines due to the down-scaling of the cross-sectional area and thus much reduced heat flow in total. This can play a critical role in determining the power and power density output of micro heat engines which was not considered in previous models.

This paper focuses on the development of a new thermodynamic model which includes both the leakage conductance and an additional source conductance (K_S) between the source and the engine emphasizing the finite heat input in real situations. Development of this model is illustrated by matching the model components to those of a representative heat engine layout. This allows the evaluation of the theoretical limits of power, power density, and thermal efficiency of micro heat engines as well as their dependence on key design parameters which can be potentially exploited for performance optimization of micro heat engines.

Table 1
Nomenclature.

T_H	hot side temperature
T_C	cold side temperature
T_S	source temperature
T_{iH}	internal hot side temperature
T_{iC}	internal cold side temperature
K_S	source thermal conductance
K_H	engine hot side thermal conductance
K_C	engine cold side thermal conductance
K_L	leakage thermal conductance
$\dot{Q}_S$	rate of heat input to system (engine and leakage)
$\dot{Q}_H$	rate of heat input to engine
$\dot{Q}_C$	rate of heat rejection from engine
$\dot{Q}_L$	rate of heat leakage
$\dot{Q}_T$	total rate of heat input
$\dot{W}$	rate of work (power)
τ	ratio of internal temperatures
k_L	material thermal conductivity of leakage path
k_S	material thermal conductivity of source
L	length of micro heat engine
A	cross-sectional area of micro heat engine
L_0	additional length due to the source conductance
A_0	additional area due to the leakage conductance
s	length scale

2. Finite heat model

Fig. 1(b) shows a proposed thermodynamic model for micro heat engines. It is composed of a source conductance K_S , a leakage component K_L , and an endoreversible engine component E , as shown inside the dashed box. The engine component E is in turn composed of a reversible Carnot engine running between a hot side temperature of T_{iH} and a cold side temperature of T_{iC} , with finite thermal conductance K_H connecting T_{iH} to the external hot side temperature T_H , and thermal conductance K_C similarly connecting T_{iC} to the external cold side temperature T_C . The leakage component is a thermal conductance K_L running between T_S and T_C , shunting heat away from the engine component. The source conductance K_S connects the source to the rest of the system, representing a finite heat input rate. The variables $\dot{Q}_S$, $\dot{Q}_H$, $\dot{Q}_C$, $\dot{Q}_L$ are cycle-time-averaged values of the heat flow rates through the conductances K_S , K_H , K_C , and K_L respectively.

The new component K_S for micro heat engines can be justified as follows. Since heat leakage is present in all engines whether large or small, it is necessary to include a heat leakage component K_L for a realistic engine model. With micro engines however, a further issue is that heat input into the engine becomes constricted as engine size is reduced, while leakage tends to increase. This causes K_L and K_S to become comparable in magnitude in micro heat engines. Whereas in a conventional engine, the magnitude of K_S would be much greater than that of K_L , and hence K_S can be ignored in the thermodynamic model. This argument is further demonstrated by considering a simplified engine layout as well as a few real micro engines.

2.1. Illustration of K_L/K_S ratio

Heat losses in real micro heat engines can be from a range of components [14,17], especially for engines with complex geometries. However, in order to illustrate the increasing K_L/K_S ratio, we consider a simplified layout of an engine chamber as shown in Fig. 2(a), to represent K_S and K_L . Note that this layout directly corresponds to the model shown in Fig. 1(b). Heat input comes externally from the top through the plate K_S and is conducted to both the engine E and the sidewalls K_L , representing the parallel heat loss through the walls of the engine. While the component E is an endoreversible thermodynamic model that has been used to model both internal and external combustion engines, including gas turbines [18], Stirling engines [19], and other engine types [16], the K_S component most accurately models heat transfer from an external source (i.e. an external combustion type engine). As will be discussed later, K_S for other engine types may be modeled differently.

The magnitude of the heat conductances mainly depends on the thermal conductivity of the engine construction material, the engine cross-sectional area and the length. Using a linear model of heat conduction, the conductances K_S and K_L can be calculated as follows:

$$K_S = k_S \frac{A + A_0}{L_0} \quad (1)$$

$$K_L = k_L \frac{A_0}{L} \quad (2)$$

Where the parameters k_S and k_L are the thermal conductivities of the source and leakage sections of the heat engine, respectively. A_0 is the cross-sectional area of the leakage path, while L_0 is the thickness of the source conductance path. A and L are the overall area and length of the heat engine. One way to investigate the effect of scaling is through an abstract length scale parameter s which represents size

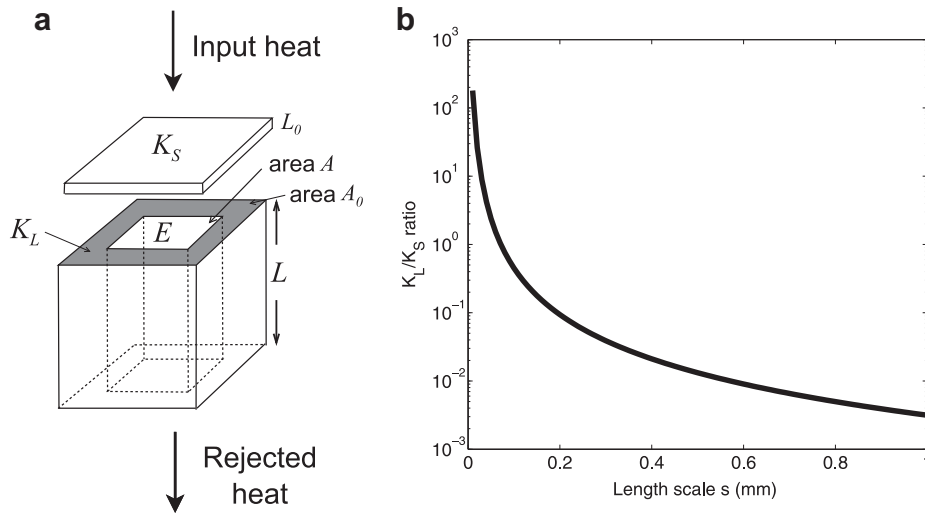


Fig. 2. (a) Illustration of thermal conductances in a micro heat engine (b) Effect of length scale on K_L/K_S ratio.

in one dimension. Thus, length L is proportional to s and area A is proportional to s^2 . From Eqs. (1) and (2), it can be seen that down-scaling of micro heat engines lead to much reduced K_S ($K_S \propto A \propto s^2$) and increased K_L ($K_L \propto L^{-1} \propto s^{-1}$). In reality, A_0 would also somewhat scale with s (e.g. if the walls of the engine were kept to constant thickness), so for the example layout this leads to a ratio K_L/K_S which is proportional to s^{-2} as shown in Fig. 2(b). The increasing K_L/K_S ratio with the down-scaling of micro heat engines suggests that heat leakage and finite heat input effect becomes relatively more prominent at the micro scale in comparison with that of macro engines and thus should be taken into consideration for modeling, which is underlined by the developed new model in this work.

2.2. Applicability to real micro engines

The P3 engine is a frequently reported [7] [20] micro heat engine that features a thermal switch to control heat addition and rejection. The P3 engine is basically a chamber filled with a phase change fluid, one wall of which is a flexible membrane that acts as the generator. Heat input into the engine is controlled by a thermal switch. Heat loss occurs through the walls of the engine as well as through the working fluid.

In the case of the P3, K_S is clearly the thermal conductance of the thermal switch, i.e. the inverse of the thermal resistance (3.8 K/W) of the switch in its “on” state [20], i.e. $K_S = 0.263$ W/K. Clearly K_S for the P3 has the same area dependence as in Eq. (1). K_L on the other hand can be calculated from the geometry and material properties using Eq. (2). The reported [2] P3 engine has cross-sectional area $A = 19.6$ mm², enclosing a cavity that is 75 μ m thick with sidewalls that are composed of layers of semiconductor tape. A working fluid, 3M PF-5060DL with thermal conductivity 0.057 W/m-K [21] is enclosed within the cavity. Due to the relatively small heat transfer area of the thermal switch (4 mm square), the heat leakage is dominated by heat transfer through the working fluid. The value of K_L can thus be calculated from Eq. (2) using the cross-sectional area and conductivity of the working fluid, the value of which is found to be $K_L = 0.015$ W/K, which leads to $K_L/K_S = 0.057$.

In comparison, the K_L/K_S ratio of a conventionally sized engine such as the GPU-3 can be roughly estimated from its published specifications [22]. K_L can be approximated using Eq. (2), from the regenerator area of about 0.01 mm² per cylinder and length of 22.6 mm. K_S is in turn approximated using Eq. (1), from the heater area of about 286.5 mm² per cylinder and length of 245.3 mm. Both

heater and regenerator are made of stainless steel, with thermal conductivity 15 W/m-K. This leads to a K_L/K_S ratio of 0.00038, which is much smaller than that of the P3 example. Hence, finite heat input is not a strong consideration for the GPU-3 engine and the new model's prediction of its performance would not differ greatly from previous models.

While not quantified, K_H and K_C in Fig. 3 represent the effective heat transfer coefficient into and out of the working fluid, respectively. These values dictate the absolute level of power that the engine is capable of generating, as in the conventional model of Fig. 1(a). The relationship of K_S and K_L on the other hand, which is the focus of this study, represents a penalty to the generated power of an engine as it is miniaturized.

For additional illustration, we consider another engine type, the MIT micro turbine [4]. K_S can be interpreted as a combustion effectiveness factor. If this micro turbine were of external combustion type, then K_S would simply be the conductance of the input heat exchanger and can be represented by Eq. (1). Since it is in fact powered by internal combustion, the geometric dependence of

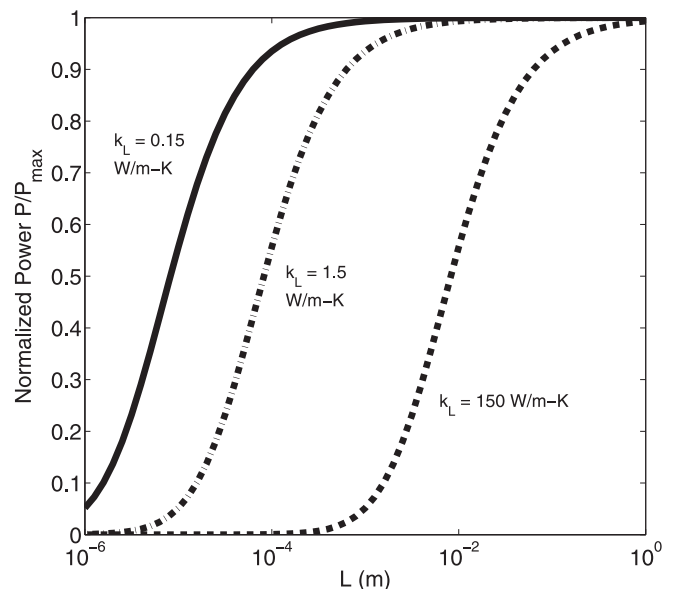


Fig. 3. Power vs engine length at different thermal conductivities, $T_H = 400$ K, $T_C = 300$ K, P_{max} calculated as power generated without leakage

K_S for this engine would be more complicated. For example, is influenced by parameters such as the chamber size, quenching distance, and fluid flow of fuel, all of which tends to reduce effectiveness of combustion as engine size is reduced [1,5]. K_L again is the heat loss through the engine walls, and can be approximated using Eq. (2), the net effect of which is that the K_L/K_S ratio tends to increase in the micro scale, which is the same result demonstrated in previous illustrations and hence justifies the use of the new thermodynamic model for micro engines.

Clearly, the developed model is most suitable for representing external combustion type engines, but can also apply to internal combustion engine types with suitable modification of Eqs. (1) and (2).

2.3. Model equations

Power and power density are key performance considerations for micro heat engines. Based on the new thermodynamic model, the theoretical limit of these performance objectives can be evaluated with consideration to leakage as follows:

The heat flow rate $\dot{Q}_H$ represents the part of heat input which is usefully converted to work, while $\dot{Q}_L$ represents the leakage that dissipates across K_L . $\dot{Q}_S$ is the total input heat rate from the source, and is the sum of $\dot{Q}_H$ and $\dot{Q}_L$.

$$\dot{Q}_S = \dot{Q}_H + \dot{Q}_L \quad (3)$$

From Fourier's law of heat conduction:

$$\dot{Q}_S = K_S(T_H - T_S) \quad (4)$$

$$\dot{Q}_H = K_H(T_S - T_{iH}) \quad (5)$$

$$\dot{Q}_L = K_L(T_S - T_C) \quad (6)$$

Substituting (4), (5), and (6) in (3), we obtain:

$$T_{iH} = \frac{(K_S + K_H + K_L)T_S - K_S T_H - K_L T_C}{K_H} \quad (7)$$

Since the internal Carnot engine is reversible, the ratio τ between the internal operating temperatures, T_{iC} and T_{iH} can be calculated as:

$$\tau = \frac{T_{iC}}{T_{iH}} = \frac{Q_C}{Q_H} = \frac{K_C(T_{iC} - T_C)}{K_H(T_S - T_{iH})} \quad (8)$$

Which leads to the expression for T_{iH} as:

$$T_{iH} = \frac{\tau K_H T_S + K_C T_C}{\tau(K_H + K_C)} \quad (9)$$

Joining Eqs. (7) and (9), T_S can be obtained as:

$$T_S = \frac{\tau(K_H + K_C)(K_S T_H + K_L T_C) + K_H K_C T_C}{\tau(K_S K_H + K_S K_C + K_H K_L + K_C K_L + K_H K_C)} \quad (10)$$

The power output can be obtained from thermal efficiency and heat input as:

$$\dot{W} = (1 - \tau)\dot{Q}_H = (1 - \tau)K_H(T_S - T_{iH}) \quad (11)$$

by inputting T_S from (10) and T_{iH} from (9), we obtain:

$$\dot{W} = K_H K_C \frac{K_S T_H + 2K_L T_C + K_S T_C - \tau(K_S T_H + K_L T_C) - \frac{1}{\tau}(K_S + K_L)T_C}{K_S K_H + K_S K_C + K_H K_L + K_C K_L + K_H K_C} \quad (12)$$

Eq. (12) is an expression of power output $\dot{W}$ as a function of the temperature ratio τ . The maximum of a function occurs at the point when its derivative is zero. Hence, by applying the condition $\frac{\partial \dot{W}}{\partial \tau} = 0$, we can obtain the maximum power that can be generated by a micro heat engine with certain leakage and finite heat input:

$$\dot{W}_{max} = K_H K_C \frac{(\sqrt{K_S T_H + K_L T_C} - \sqrt{K_S T_C + K_L T_C})^2}{K_S K_H + K_S K_C + K_H K_L + K_C K_L + K_H K_C} \quad (13)$$

with the following expression for τ corresponding to the maximum power point:

$$\tau = \sqrt{\frac{(K_S + K_L)T_C}{K_S T_H + K_L T_C}} \quad (14)$$

The relevant power density can thus be obtained as:

$$\text{Power density} = \frac{\dot{W}}{\text{total volume}} = \frac{\dot{W}}{(A + A_0)(L + L_0)} \quad (15)$$

By using Eqs. (13) and (15), the effect of the leakage length and thermal conductivity on the limit of maximum power and power density of micro heat engines can be obtained, which can then provide practical design guidance. For instance, since the maximum power is dependent on various thermal conductances within the heat engine, it suggests that the development of micro scale heat exchangers such as the P3 thermal switch [20] could have a large effect on the power output. It is important to note that in extreme and ideal cases when $K_S = \infty$ or when $K_L = 0$, Eq. (14) is reduced to $\tau = \sqrt{T_C/T_H}$, which is the maximum power condition predicted by using the previously reported thermodynamic model in which infinite heat input was implicitly assumed [23].

Another quantity of interest is the thermal efficiency at maximum power η , which can be expressed as the maximum power $\dot{W}_{max}$ over the total heat input rate $\dot{Q}_S$:

$$\eta = \frac{\dot{W}}{\dot{Q}_S} = \frac{\dot{W}}{K_S(T_H - T_S)} \quad (16)$$

With the value of $\dot{W}_{max}$ obtained from (13) and T_S from (10).

3. Analysis and discussion

3.1. Power

Since power generation is one of the key applications for micro heat engines, it is important to predict and optimize the upper limit of maximum power that can be potentially produced. Design parameters such as thermal conductivity of the engine construction material, length scale, and hot side temperature can all play important roles, as indicated by Eq. (13). Fig. 3 shows the power output as a function of engine length L . It is observed that for larger length scales, the power output is close to P_{max} which represents power generated without any leakage. However, with the decrease of L , the power output rapidly falls off due to the increase of leakage and K_L/K_S ratio. The minimum L for a viable device depends largely on the thermal conductivity k_L . For example, to avoid excessive leakage, silicon ($k_L = 150$ W/m-K) is only suitable for engines with L in the range of 10–1000 mm; whereas for smaller engines with L in the range of 0.1–1 mm, polymers such as PMMA and SU-8 ($k_L = 0.1$ – 0.2 W/m-K) are better choices as construction materials. Fig. 4 (a) shows normalized power vs. length at different source temperatures (T_H) where output power increases at higher source temperatures. If the micro heat engine is designed for an output

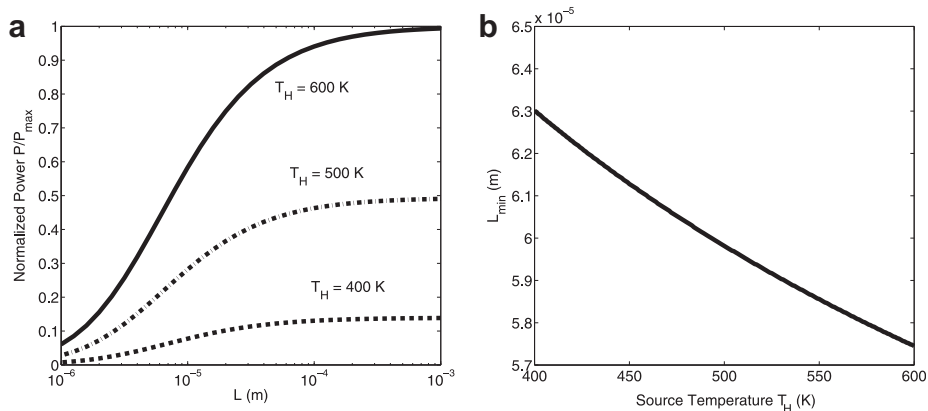


Fig. 4. (a) Power vs engine length at different source temperatures, P_{max} calculated as power generated without leakage. (b) L_{min} at different source temperatures. Values calculated at $T_C = 300$ K, $k_L = 0.15$ W/m-K.

power level as a percentage of P_{max} , for example, to 90% of P_{max} , then for a given T_H and k_L , the L_{min} to achieve this power output can be determined from Eq. (12), a typical result of which is shown in Fig. 4(b) for $k_L = 0.15$ W/m-K.

Figs. 3 and 4 suggest that in order to increase the theoretical upper limit of power output at a given T_H , L_{min} and k_L should be taken into consideration during the engine design stage. Taking the P3 as an example again, based on the reported dimensions as well as K_S and K_L values calculated previously, a theoretical W_{max} of 9.7 mW can be obtained using Eq. (13). In reality a power output of 0.35 mW was reported [2]. Despite possible errors in the estimation of K_S and K_L values, it is still notable that there seems to be plenty of room for possible performance improvement in P3 engines. For example, the power output of P3 engines have been further improved to 2.5 mW with the implementation of active cooling [2]. Alternatively, improving the thermal efficiency of the thermodynamic cycle, e.g. by means of a regenerator, could help the engine to achieve power generation closer to its thermodynamic maximum. In addition, the model can be used to evaluate the relative effects of design changes, e.g. adjusting L or k_L to reduce leakage, for example by increasing the P3's length from 75 μ m to 200 μ m, the theoretical limit of power output predicted by this model would increase from 9.7 mW to 10.4 mW, a 7% improvement. The relatively small improvement of this design change suggests that heat leakage is not a major concern in the current designs of the P3.

3.2. Power density

For power applications where mobility, weight, and space are major concerns, e.g. as a portable power supply, the power density is a key performance objective [1] and thus is discussed in this section. The theoretical upper limit of power density is calculated as the maximum power divided by the volume, as given in Eq. (15). Since weight is generally proportional to volume, the power density calculated here is taken to be the equivalent of power per unit weight.

Fig. 5 shows the power density (normalized against the maximum power density reachable) which shows that as the length scale is reduced, power density increases until it reaches an optimum value and then decreases due to the domination of the heat leakage. Thus, to aim for high power density with a given engine material conductivity (k_L) and source temperature (T_H), it is important to design an engine with an optimum length. However, in the case of variable T_H , the optimum length L_{opt} is a function of T_H , as shown in Fig. 6 (a). The optimum lengths can be calculated from Eq. (14) and a set of typical results is shown in Fig. 6 (b).

For the case of the P3 engine, the model predicts that the theoretical limit of power density is 6.6 mW/mm³ for the actual engine length reported ($L = 75$ μ m). Since the actual density reported is only 0.24 mW/mm³, as noted in the power discussion, further development is needed to bring real power and power density closer to these upper limits. Nonetheless, the relative effects of a design change can be evaluated by means of the model. Based on the reported operating conditions of the P3 ($T_H = 333$ K, $T_C = 300$ K, and $k_L = 0.057$ W/m-K), the optimal length for maximum power density can be calculated to be 4 μ m, which corresponds to a theoretical limit of maximum power density of 33.9 mW/mm³. We acknowledge that practical fabrication issues might limit the realization of such dimensions and power density.

3.3. Thermal efficiency

Thermal efficiency for micro heat engines has been the focus of studies in previous models. Other factors affect the overall system efficiency of micro heat engines, such as mechanical friction and fluid-dynamic losses, however only the thermal efficiency is considered in this work.

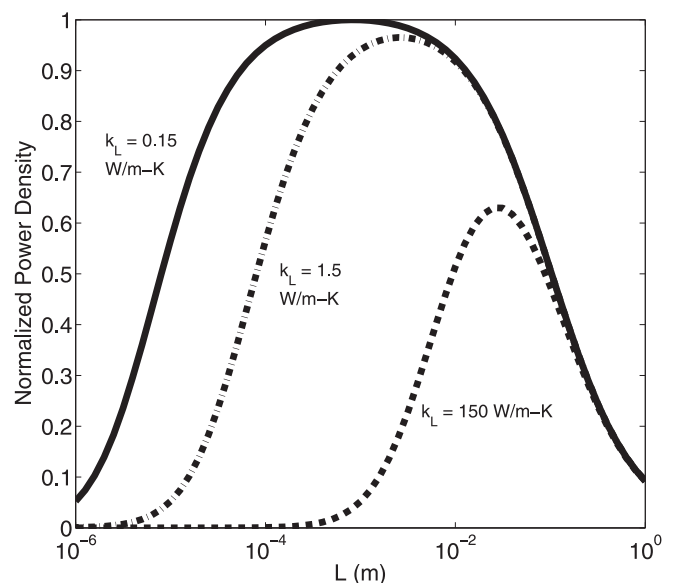


Fig. 5. Power density vs length scale at different thermal conductivities, power density normalized to value without leakage. Values calculated at $T_H = 400$ K, $T_C = 300$ K.

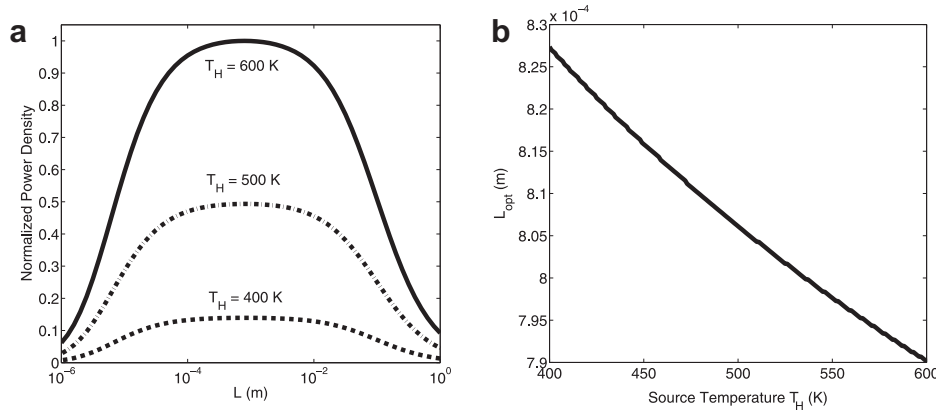


Fig. 6. (a) Power density vs length scale at different source temperatures, power density normalized to value without leakage. (b) Optimum lengths at different source temperature. Values calculated at $T_C = 300$ K, $k_L = 0.15$ W/m-K.

Generally speaking, the increased heat leakage due to down-scaling of micro heat engines leads to a decrease in thermal efficiency. Although relatively low efficiency micro heat engines can still find practical applications [24], it is important to improve efficiency especially for power conversion applications [2].

The upper limit of thermal efficiency for any heat engine is the Carnot efficiency, but it has been shown from heat engine models such as the one in Fig. 1 (a) that the thermal efficiency at maximum power is somewhat lower than Carnot efficiency [16]. The thermal efficiency at max power however has been shown to correspond better with real power plant data [14], and hence may be more realistic to use for performance evaluation. The effect of length L on the thermal efficiency at maximum power η can be obtained from (16), and a set of typical results is shown in Fig. 7.

For a given K_L , the upper limit of thermal efficiency at maximum power can be found at large values of L . Alternatively, since thermal efficiency decreases with L , the lower limit of L can be found for a targeted efficiency. Note also that choosing a material with lower thermal conductivity k_L allows a lower value for L , which agrees with the result from previous modeling work [11]. For the case of the P3 engine, with $T_H = 333$ K, $T_C = 300$ K, and $k_L = 0.057$ W/m-K, the thermal efficiency at maximum power achievable is 1.47% as

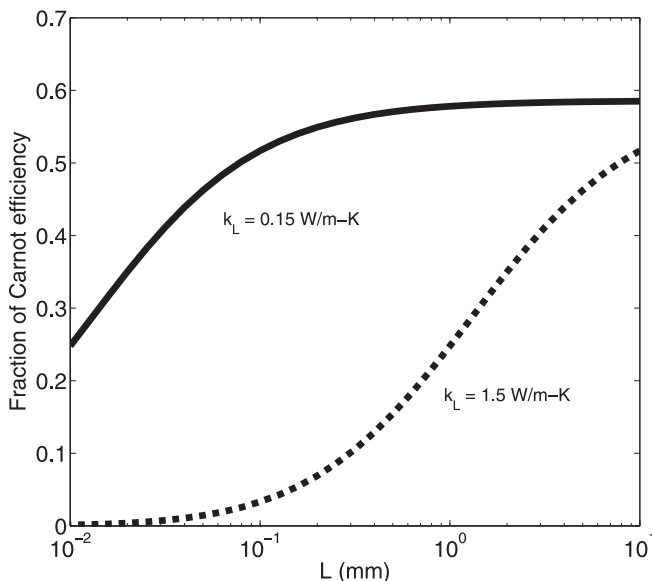


Fig. 7. Thermal efficiency vs Length, thermal efficiency normalized against Carnot efficiency. Values calculated at $T_H = 400$ K, $T_C = 300$ K.

calculated by this model, while the actual thermal efficiency of 0.2% is reported [2]. Tailoring of k_L or L can potentially improve the thermal efficiency. For example, if the design length of the P3 were changed from 75 μ m to 200 μ m, the theoretical efficiency can be potentially improved from 1.47% to 2.7%.

3.4. Design trade-off for performance

Ideally in micro heat engine design, the designed performance should include high power, high power density, and high efficiency, but in practice micro heat engine performance may involve tradeoffs among these objectives. The trade-off between power and thermal efficiency is known for general heat engines from endoreversible thermodynamics [16]; that is, at maximum Carnot efficiency the engine is operating infinitely slow, hence generating zero power. Maximum power is generated at a thermal efficiency somewhat below the Carnot efficiency. However in real heat engines with leakage, it has been shown that the maximum power point and maximum thermal efficiency point are in close proximity [15], and can practically be treated as equivalent. Hence, maximizing power does tend to maximize thermal efficiency, as Eq. (16) shows.

Another trade-off that needs to be considered for micro heat engines is that between power and power density. It is important to note that the optimum length for maximum power density is different from that for maximum power. The L for maximum power is relatively large in order to minimize leakage, however this large L (i.e. large volume) leads to reduced power density. Hence, reduced power output is obtained when the optimum length for power density is designed. One possible way to benefit from high power density is to use a large array of micro heat engines with high power density to collectively produce high overall power output collectively, which would be greater than that of a single engine with similar overall volume if losses from interconnection of such an array is ignored. Thus, with optimized designs, micro heat engine arrays could be collectively used to generate high power output if they can be potentially assembled into large areas or volumes.

4. Conclusion

A new thermodynamic model was developed to evaluate the theoretical maximum power output, power density, and thermal efficiency of micro heat engines, which includes both heat leakage and finite heat input elements. By analyzing this model, it was discovered that as the length of heat engines are reduced to micron-scale, the theoretical limit of power output generally

decreases. In order to improve power output, a lower limit to the micro engine length L can be identified for a given set of design parameters such as temperature and geometric dimensions; similarly, suitable materials can be selected to reduce detrimental heat leakage. Furthermore, the theoretical limit of power density was evaluated. For a given engine material (k_L) and source temperature (T_H), an optimal length can be found that maximizes the limit of power density. The theoretical limit of thermal efficiency at maximum power for micro heat engines was also studied and found to trend similarly to power output. Finally, in aiming for ideal overall performance, design trade-offs for power, power density, and thermal efficiency were examined, possibly leading to micro heat engine arrays that have higher overall output power than a single engine with equivalent size. The predictions of the developed model have been compared to a typical micro heat engine throughout the studies of theoretical limits of power, power density, and thermal efficiency. By comparing these results, potential design improvements were discussed.

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